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CERTIFICATE

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I hereby certify that annexed is a true copy of the Provisional Specification as filed on 15 July 1999 with an application for Letters Patent number 336765 made by CHRISTOPHER FREDERICK BAYNE.

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Dated 2 August 2000.

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PROVISIONAL SPECIFICATION

Title: Seal for Powdered Materials

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do hereby declare this invention to be described in the following statement:

There are countless applications in industry in which a mechanism is called for which comprises a rotatable member (such as a shaft) which passes through or is seated or supported in an aperture in some other part of the mechanism (such as a wall or bearing housing) which is stationary relative to the rotatable member or at least does not rotate therewith.

The present invention is particularly but not necessarily exclusively concerned with equipment in which such mechanisms operate in an environment in which finely divided solids are present, the solids being carried either in a fluid such as air (or other gas) or water (or other liquid). Such equipment includes screw conveyors, bucket elevators, ribbon mixers, so called Z mixers, rotary valves, augers, pulverizers, hammer mills, powder transfer equipment and other mixing equipment. This list is not intended to be exhaustive and the invention could in principle be applied to any suitable equipment which operates in conditions in which it is desirable that gas or liquid borne solid material should be excluded from bearings or other components of the equipment or should, for other any other reasons, including environmental, health or safety reasons, be contained within a vessel or other enclosed space.

In the case of equipment comprising a rotatable member/aperture combination as described above, seals are used to prevent liquid, gas or solid matter (or a mixture thereof) from passing into or through the aperture. A seal often comprises a part which is fixed on the rotatable member. This part of the seal is commonly called a rotor. Similarly, a seal often comprises a part which is fixed on the stationary part of the mechanism. This part of the seal is commonly called a stator.

Every such seal must inescapably comprise an interface at which the rotor contacts or comes close to the stator and careful attention must be paid to the design of the components at the interface. If any leakage of the fluid medium in which the seal is working takes place past the seal, such leakage is most likely to occur at the interface. Also, a primary cause of failure of a seal is due to abrasive matter which may be carried in the fluid medium which

penetrates between the stator and the rotor at the interface or the build up of heat due to rubbing contact between the stator and the rotor at the interface.

The present invention relates to so called labyrinth seals. This term is used somewhat loosely in the art but, for the purposes of this specification, a labyrinth seal is one in which, by design, there is no intentional contact between the rotor and the stator at the interface. Rather, the parts of the seal are shaped so that there is a fine and usually tortuous passage (or 'labyrinth') extending along the interface between the rotor and the stator. In conventional labyrinth seals which are designed for use with liquid working media the passage is commonly so narrow that, by design, only a thin film of the liquid is present in the passage. This film is adherent to the surfaces of the stator and the rotor along the interface and it is the film which, in principle, prevents the liquid working medium from passing through the passage when the seal is in use.

In conventional labyrinth seals which are designed for use with gaseous working media, the passage is commonly even narrower than the passage in a seal for liquids. The parts of the seal (including the parts of the stator and the rotor which form the passage at the interface) are arranged to bring about a pressure gradient between the ends of the passage which, in principle, prevents the working medium from passing through the passage when the seal is in use.

Various means have been used to increase the effectiveness of conventional labyrinth seals. For example, most seals for liquid working media have some means for removing any liquid which penetrates more than a certain distance into the passage. Many seals (commonly called dynamic seals), for both liquid and gaseous working media, have some means for raising the ambient pressure in the passage (for example by pumping liquid or gas into or through the passage) to a level which is higher than the pressure of the medium in which the seal is working. The stators of yet other seals are provided with packings, brushes or the like which do in fact make contact with the rotors in use. Packings are often used where there is a need to prevent abrasive material carried in the working medium from penetrating to the

interface. So-called brush seals are commonly used in turbine engines.

The conventional labyrinth seals of which the applicant is aware are generally intended for use in mechanisms where the shaft or other rotating member on which the rotor is mounted rotates at relatively high speed. Indeed, in many cases, it is essential that the seals be operated at high rotational speeds if they are to function properly. By contrast, the shafts in material handling and processing equipment of the type described above generally operate at relatively low rotational speeds. For this reason, and also due to the fine clearance between the parts of the stators and the rotors of conventional seals, the latter would be unsuitable to be used in applications where there is present a substantial amount of abrasive material such as grit or dust in the working medium. There are however many industrial applications which call for such seals and it is well known that the seals which are conventionally used in such applications are not very satisfactory.

It is one object of the invention to provide a labyrinth seal which might find use in at least some applications in which the working medium is a dust producing pulverulent solid or a gas or liquid in which a substantial amount of finely divided solid material is suspended.

According to one aspect of the invention, there is provided a seal for use in an environment comprising a fluid borne pulverulent solid, the seal comprising at least one pair of elements including a first element and a second element one of which constitutes a stator and the other of which constitutes a rotor which, when the seal is in use, is rotatable with respect to the stator, the first element being provided with an annular projection which in use projects into an annular recess formed in the second element, the annular recess and the annular projection having interfacing surfaces between which there is clearance, the arrangement being such that, in use, layers of the pulverulent solid can be deposited on the interfacing surfaces to define a passage extending along the interface, the width of the passage being substantially the smaller than the clearance between the interfacing surfaces.

According to another aspect of the invention, in a mechanism which operates in an

environment comprising a fluid borne pulverulent solid and which comprises a first part which rotates relative to a second part, there is provided a method of providing a seal between the two parts, the seal comprising at least one pair of elements including a first element and a second element one of which constitutes a stator mounted on the first part and the other of which constitutes a rotor mounted on the second part, the first element being provided with an annular projection which in use projects into an annular recess formed in the second element, the annular recess and the annular projection having interfacing surfaces between which there is clearance, the arrangement being such that, in use, layers of the pulverulent solid can be deposited on the interfacing surfaces to define a passage extending along the interface, the width of the passage being substantially the smaller than the clearance between the interfacing surfaces.

In one form of the invention, the fluid is a gas. In an alternative form of the invention the fluid is a liquid.

The effectiveness of the seal is determined by the width of the passage between the layers of solid material. This passage constitutes a labyrinth and the operating conditions in the equipment and the layout of the passage should thus be conducive to allowing the layers to be formed and to be stable once they are formed. The applicant has found that, provided the layout and dimensions of the passage are such as to ensure that the rate of flow of the fluid through the passage before the layers begin to be formed is small enough to ensure that solid material is deposited on the interfacing surfaces by such fluid, the layers are surprisingly stable once they have been formed. The rate of flow in turn is determined largely by the length of the passage and, because the clearance between the interfacing surfaces in the passage is large compared to conventional labyrinth seals, the length of the passage of a seal falling within the ambit of the invention will usually be relatively great. According to one aspect of the invention, the minimum value of the ratio of the length of the passage to the smallest diameter of the annular passage is not substantially less than 1:2 or 50%. In the seal assemblies of the invention this ratio will however commonly be substantially greater than this. In the some of the embodiments described herein with reference to the drawings, the

ratio is greater than 1 and in one case nearly equal to 2.

Providing that the passage is tortuous also helps to achieve the desired low rate of flow of fluid through the passage.

Another means of ensuring that the solid material is deposited on the interfacing surfaces is by providing that the width of the interfacing surfaces is relatively large compared to conventional labyrinth seals. According to another aspect of the invention, therefore, the minimum value of the ratio of the width of the interfacing surfaces to the smallest diameter of the annular passage is not less than 1:20 or 5%. In the seal assemblies of the invention this ratio will however commonly be substantially greater than this. In the some of the embodiments described herein with reference to the drawings, the ratio is greater than 1:5.

Although the interfacing surfaces may in principle be disposed at any angle to the rotational axis of the rotor, manufacturing considerations are likely to be the determining factor in selecting the angle of disposition and in most cases the interfacing surfaces will be substantially perpendicular or parallel to the rotational axis. It is also thought that the seal is most effective when the interfacing surfaces are substantially perpendicular to the rotational axis.

There are many other factors which might have an influence on the stability of the layers. These factors include the nature of the fluid and the solid material and the particle size and moisture content of the latter; the operating conditions of the equipment such as the size and rotational speed of the shaft and the pressure of the fluid. The experimental work carried out by the applicant to date however suggests that once a seal has been found to work satisfactorily in a given application (especially in the materials handling applications suggested above) the same seal is likely to perform satisfactorily despite quite wide variations in these other factors.

The solid material could, for example, include dry powders, dust, granules, sand, grit, ore

and crushed rock any of which, as is well known, produce substantial quantities of grit and air borne dust when they are being handled or treated in large quantities in industrial applications. These examples of solid material could also occur in water or slurries of other liquids.

The invention is further discussed with reference to the accompanying drawings in which:

Figure 1 is a cross sectional side view of one example of a first seal assembly according to the invention mounted on a rotatable shaft;

Figure 2 is an end view of a split ring, being one of the components of the seal assembly shown in Figure 1;

Figure 3 is an enlarged cross sectional side view, showing parts of the first seal assembly separated from each other;

Figures 4 to 8 are cross sectional side views each of a seal assembly according to the invention mounted on a rotatable shaft.

For the sake of avoiding repetition, in this specification the use of the phrase 'in the present example' or words to the same effect is intended to indicate that what is being described is by way of illustrative example and that the scope of the invention is not intended to be limited thereto unless a contrary intention appears from the context. On the other hand, in the absence of a phrase such as 'in the present example' or words to the same effect, it should not be taken that the scope of the invention is to be limited by any matter described unless it is clear from the context that this is intended.

Referring first to Figures 1 to 3 there is shown a shaft 10 carried in a ball bearing assembly 12 seated in housing 14. The ball bearing assembly 12 and the housing 14 can be a commercially available 'off the shelf' unit and need not be described in detail. The housing

14 is mounted over four threaded machine bolts or studs 16 fixed on a plate 18 and held in place by means of nuts and washers 20.

In the present case the plate 18 is the vertically disposed end wall of a vessel constituting the casing of a screw conveyor. The shaft 10 carries a helical flight arrangement by means of which material which is fed into one end of the screw conveyor is transported through the vessel. Screw conveyors are typically required to handle comminuted solid materials which are either in dry form or mixed with water or other liquid to form slurry. The shaft 10 is arranged to be driven by (for example) an electric motor through a reduction gear box coupled through a conventional coupling to the shaft. Only the shaft 10 of the screw conveyor is shown since the other aforementioned components thereof are of conventional design.

It should be understood that the plate 18 and the other components shown in the drawing are not limited to the described application and might in fact constitute any component of an open or closed vessel which is part of any suitable equipment including materials handling equipment of the type discussed above. Furthermore, in such other application the plate 18 need not be vertically disposed. It might be disposed horizontally or at any angle between horizontal and vertical.

When the shaft 10, mounted in the bearing 12 and housing 14, is located in the position shown in the drawing, the shaft projects through an aperture 22 in the plate 18 with the rotational axis 24 perpendicular to the plate 18. In the orientation shown in the drawing, the interior of the vessel is to the right of the plate 18. Whether the material being handled by the equipment is wet or dry, it is necessary to protect the bearing from abrasive matter entrained in such material and a seal assembly 26 is provided for this purpose.

The seal assembly comprises a rotor 28 (see Figure 3) fixed on the shaft 10. The rotor may be machined from a work piece of stainless steel or any other suitable material. The rotor is annular, having an axis which is concentric with the rotational axis 24 of the shaft. The rotor

has a bore 30 through which the shaft 10 projects. Two O-ring sealing elements 32 are located in channels formed in the face of the bore 30. The elements 32 prevent leakage of the medium handled by the screw conveyor past the interface between the bore 30 and the shaft 10.

The rotor comprises a hub 34 with two mutually identical fins 36, 38 formed integrally with the hub. Each fin 36, 38 has opposed, highly polished radial faces 40a, 40b which, in the present example, are parallel to each other and are disposed perpendicular to the rotational axis 24. The fins 36, 38 are spaced from each end of the hub 34 and, with the outer face of the hub, define annular recesses 42, 44 located adjacent the respective ends of the hub. The fins are also spaced from each other so that, with the outer face of the hub, they define an annular channel 46 located between the fins.

The seal assembly further comprises a stator assembly 50. The stator assembly includes a retaining plate 52 and a number of annular elements or rings 60, 62, 64. In the present example, the shape of the axial periphery of the retaining plate 52 is substantially identical to that of the bearing housing 14 and the retaining plate is provided with four holes through which the studs 16 pass. The radial faces 51, 53 of the retaining plate are machined so that the thickness of the retaining plate is accurately controlled. In use the retaining plate is clamped between the bearing housing 14 and the machined outer face 57 of the plate 18 by means of the nuts 20.

A bore 58 is machined in the retaining plate 52. The bore is concentric with the rotational axis 24 of the shaft. The rings 60, 62, 64 are inserted in the bore 58. The ring 60 is generally L-shaped, comprising an axially extending portion which will convenience be called a sleeve 66 and a portion which will be called a flange 68 which extends radially inwardly from one end of the sleeve 66. The sleeve is a press fit in the bore 58 and the flange is located in use in the annular recess 44 of the rotor with the outer face 69 of the flange 68 flush with the right hand outer face 51 of the retaining plate 52 and against the wall 18. The overall width of the ring 60 is substantially equal to the thickness of the retaining plate 52 so that when the

flange 68 is located against the wall 18, the free end 65 of the sleeve 66 is substantially flush with the outer face 53 of the retaining plate 52. The bore of the sleeve 66 is divided into two portions 72 and 74 by a step 70. The portion 72 is located between the flange 68 and the step 70. The diameter of the bore of the portion 72 is smaller than that of the portion 74. A recess 77 is formed in the inner face of the flange 68. This recess accommodates an annular setting plate 79, the function of which is discussed further below.

The ring 62 is also L-shaped, having an axially extending sleeve 76 and a flange 78 which extends radially inwardly from one end of the sleeve 76. The sleeve 76 is a press fit in the bore of the portion 72 of the ring 60 and the overall width of the ring 62 along the sleeve 76 is substantially equal to that of the bore of the portion 72 so that, when the ring 62 is in place, the outer face 80 of the flange 78 is flush with the step 70.

When the seal is assembled, the flange 78 of the ring 62 is accommodated in he annular channel 46 of the rotor 28. The method of assembly of the seal assembly 26 is discussed below but it is convenient to mention here that, in order to be able to assemble the seal, the ring 62 is 'split'. That is, the ring is divided in the present example into two semi-annular segments 62a, 62b which are separated at the radially extending interfaces 81 located substantially on a diameter of the ring 62.

The ring 64 is also L-shaped, having an axially extending sleeve 82 and a flange 84 which extends radially inwardly from the outer end of the sleeve 82. The sleeve 82 is a press fit in the bore of the portion 74 of the ring 64 and the width of the ring 64 along the length of the sleeve 82 is substantially equal to that of the bore of the portion 74 so that, when the ring 64 is in place, the outer face 86 of the flange 84 is flush with the end 65 of the sleeve 66. A recess 88 is formed in the inner face of the flange 84. This recess accommodates a second annular setting plate 90 which is identical to the setting plate 79.

When the seal has been assembled the flange 68 is located in the recess 44. The thickness of the flange 68 is such that there is clearance between the inner radial face 71 of the flange 68

and the outer face 40a of the ring 38 of the rotor. Similarly, the thickness of the flange 84 is such that, when it is located in the recess 42, there is clearance between the inner radial face 83 of the flange 84 and the outer face 40a of the ring 36 of the rotor. Furthermore, the thickness of the flange 78 is such that, when it is located in the channel 46, there is clearance between each of the radial faces of the flange 78 and the inner faces 40b of the respective rings 36, 38. There is also clearance between the inner ends of the flanges 68, 78, 84 and the interfacing parts of the outer face 92 of the rotor hub 34.

The faces of the rings between which there is the aforementioned clearance constitute a composite interface between the rotor and the stator. Because of the clearance, there is, by design, no contact between the rotor and the stator so that, along the interface, there is a passage extending from one side of the seal to the other. After the seal is assembled and before it is put into use, the clearance is substantially greater than the clearance along the interface between the rotor and stator of conventional labyrinth seals. The clearance must be large enough to allow finely divided solid material which is suspended in the fluid contained in the vessel of the screw conveyor to be carried into the passage by the fluid where it is deposited on those interfacing portions of the rotor and stator which define the passage to build up in layers. After the seal has been in use for some time the thickness of these layers increases to the point where the width of the passage is reduced to a substantial extent and eventually the layers come into contact with each other as the rotor rotates in the stator. At this point the width of the passage is so small that, effectively, none of the fluid medium can pass through the passage.

In order for the finely divided solids to be able to enter the passage and start to build up the layers, the initial width of the passage (i.e. the clearance between the rotor and the stator along the interface before the seal is put into use) must be at least double the size of the finest particles of solid material which will occur in significant quantity in the fluid. In most circumstances however, the clearance can be substantially greater than this minimum without appreciably affecting the building up of the layers or the operation of the seal. It is indeed possible that in many cases, particularly where the particle size is small, the clearance

will need to be substantially greater than twice the particle size owing to practical problems encountered in constructing and operating such seals. It is necessary for example that the design of the components make allowance for relative radial and axial movement of the components at the interface. Such relative movement occurs as the rotor rotates but can also occur, for example, due to thermal expansion, vibration and misalignment of the components. There is in particular a practical and economic limit as to how flat the radial faces of the components can be made. The less flat such radial faces are the greater will need to be the clearance therebetween between at an interface. Furthermore, it is common for the shaft on which a seal rotor is mounted to run out of true and this fact requires that both the axial and radial clearance between the components of a seal at the interface be increased. The clearance must not however be so great that the solids are unable to build up into stable layers or that such layers are liable to be damaged by particles carried into the passage by the fluid (i.e. gas or liquid) medium. Either of these problems might be exacerbated if the fluid medium is able to flow through the passage at appreciable speed. A clearance which is too great could also have the further unacceptable result that the fluid might get to the bearing or escape from the vessel to the surrounding environment before the layers start to form. The optimum clearance at an interface between the components of a seal assembly of the invention can be established by testing. In the example illustrated, which is intended for a screw conveyor handling aluminium oxide the minimum particle size of the dust of which is about 15μ, the clearance between the interfacing components of the seal in both the axial and radial directions is 0.5mm. By way of further illustration, in seal assemblies for mounting on shafts of between about 20 mm and about 50 mm diameter installed in material handling equipment of the type described above, it has been found that a suitable clearance in both the axial and radial directions between the components at an interface is about 0.5-1.00 mm.

The overall length of the passage will also have an appreciable effect on the rate at which the fluid is able to flow through the passage. In the present case the length of the passage is effectively equal to the sum of the widths of each of the two radial faces 40a, 40b of each of the fins 36, 38, the sum of the axial thickness of the fins 36, 38 and the axial distance

between the fins 36, 38. In the example illustrated this length is about 64 mm and the minimum diameter of the rotor at the interface is 58 mm. In principle, there is no upper limit to the length of the passage and the upper limit will probably be determined by cost and operational considerations. The optimum length can be established by testing in any particular case. In practice the length of the passage is unlikely to be less than 50% of the minimum diameter of the rotor at the interface at least in screw conveyors and materials handling equipment of the type described above.

It is to be noted that, in common with many conventional seals, the faces of the rotor and the stator which make up the interface are disposed either at 90° or parallel to the rotational axis of the shaft. It is thought that the fact that such faces are disposed for the most part perpendicularly (i.e. at 90°) to the rotational axis contributes to the effectiveness of seals according to the invention. It is however considered that a seal in which such faces are disposed for the most part parallel to the rotational axis or at an angle other than 90° thereto will in principle also fall into the ambit of the invention, particularly if the widths of such faces is also substantially greater than in the case of conventional seals.

It is considered in particular that the fact that the widths of such faces are substantially greater than in the case of conventional seals contributes to the effectiveness of the seal. It is considered that, in seals according to the invention at least for screw conveyors and materials handling equipment of the type described above, the optimum widths of such faces, whether disposed perpendicular or parallel to the rotational axis of the shaft, is unlikely to the less than about 20% of the shaft diameter.

In the example illustrated in Figure 1, the width of the radially extending faces 40a, 40b of the rotor (which make up the major part of the interface) is 12.5 mm which is 25% of the shaft diameter.

The length of the passage will also be affected to some extent by the pressure differential

across the seal. However, in many of the applications of the type described above for which the seal is primarily intended, it is likely that there will be no significant such pressure differential. In any case it is thought that, for practical purposes, such pressure differential should not exceed about 1 bar. However, where a significant pressure differential is likely to occur, a conventional mechanical seal may be installed downstream of a labyrinth seal according to the invention as there will be substantially no solid particles in the fluid medium which is presented to the mechanical seal.

It is not thought that the rotational speed of the rotor has a significant deleterious effect on the operation of the seal, at least at the relatively slow rotational speeds of the rotors in applications such as those described above for which the seal is primarily intended. Indeed, the build up of the layers to form the labyrinth passage as described above is probably promoted by the slow rotational speed of the rotor. In the example, the shaft size is 50 mm diameter and the designed rotational speed is 700 rpm.

For a shaft having a diameter between 25 and 65 mm, rotating at a speed of up to 1500 rpm, in a materials handling machine of the type described above in which the particle size of dry material is not greater than 100μ, it is thought that the clearance between the interfacing faces of the seal components would typically be between about 0.5 mm and 1.0 mm. Where the material is suspended in a liquid medium such as water or is damp, it is thought that the same seal would remain effective although it might have to be used in conjunction with other components as discussed further below. The optimum size of all of the working parameters may be established in each case by experiment.

The function of the setting plates 79, 90 is primarily to ensure that the correct clearance is maintained between the faces of the components when they are being assembled. The width of the setting plates is therefore selected so that, after assembly, the clearance between the face of each setting plate and the opposing face of the respective fin 36, 38 is about 0.25 mm or half of the clearance between the opposing faces of the rings 60, 64 and the fins 38, 36. The setting plates are sacrificial; i.e. it is expected that they will deteriorate rapidly after the

seal is put into service and they play little or no part in the actual operation of the seal after such deterioration occurs. However, as discussed below, analogously to the function of the lip seals 150 provided in the assembly 100 illustrated in Figure 4, the setting plates 79, 90 may have some limited effect in contributing to the build up of material on the interfacing faces of the seal immediately after the seal is put into use.

The components of the seal assembly 26 may be made of the same materials which are used for the similar components of conventional seals. Thus the rotor and the rings 60, 64 may be made an abrasion resistant material, preferably of metal. For many applications the material may also need to be corrosion resistant and/or capable of taking and retaining a high polish in which case the material may be, for example, bronze, stainless steel or a ceramic material. The split ring 62 and the setting plates 79, 90 may be of Teflon or other suitable self-lubricating synthetic plastics material which is softer than the rotor so that it will wear preferentially.

An advantage of using a material such as Teflon for the split ring 62 is that the two segments 62a, 62b may be derived from a single ring which is machined to shape and split along a diameter by the so called 'random cracking' method. This involves scoring the ring on a diameter and giving the ring sharp taps at the score lines. Alternatively a knife may be placed on the ring at a diameter and given a sharp tap. In these circumstances the ring will break apart into the two segments along the score lines. The interfaces between the two segments at the breaks will be jagged which helps to lock the two segments together in the correct relative position in use.

If the split ring is made of metal, a useful method of splitting the single ring is by means of so-called wire cutting or spark erosion. An advantage of this method is that the cut which is made in the material in the splitting operation is very fine so that the splitting operation can be carried out after the original single ring is finish machined, the two segments 62a, 62b formed from the original ring both being usable. Another advantage of this method is that the pattern of the cut can also be jagged as indicated at 81' which, again, helps to lock the

two segments together in use.

The split ring may also of course be formed by other conventional methods such as by finish machining two ring halves placed together after the ring has been partially machined.

The setting plates may also be of alternative materials known to be suitable for the purpose including hardened and ground tool steel, or oil-, or carbon-filled nylon.

One advantage of the seal assembly 26 is that the retaining plate 52 may be made of a relatively inexpensive material such as cast iron or aluminium. In conventional seals, the part of the stator in which a split ring is inserted is formed as a unitary body made of bronze, stainless steel or the like. This adds to the cost of conventional seals.

The seal assembly 26 is sold as an 'off the shelf' unit with the components having been assembled together in the following manner. The ring 60 is first pressed into the bore 58 of the retaining plate 52 until the outer face 69 of the flange 68 is flush with the outer face 51 of the retaining plate. The setting plate 79 is inserted in the recess 77 of the flange 68. The two segments 62a, 62b of the ring 62 are inserted in the channel 46 of the rotor 28 and the ring 62 (together with the rotor 28) is then pressed into the bore 72 of the ring 60 until the free end 75 of the sleeve 76 abuts the inner face 67 of the flange 68. At this stage, the outer face 80 of the flange 78 is flush with the step 70. The setting plate 90 is inserted in the recess 88 of the ring 86 which is then pressed into the bore 74 of the ring 60 until the end 63 of the sleeve 64 abuts the step 70. At this stage the outer face 86 of the flange 78 is flush with both the left hand end 65 of the sleeve 66 and the face 53 of the retaining plate.

To mount the seal assembly, any coupling flange, pulley, bearing or the like mounted on the portion of the shaft which protrudes out of the vessel through the aperture 22 must first have been removed. The rotor 28 is mounted over the free end of the protruding portion of the shaft 10 and pushed along until the retaining plate comes into contact with the wall 18 of the vessel. The bearing 14 is then mounted in the same way and fixed in place by tightening the

nuts 20 with the seal assembly sandwiched between the bearing and the wall 18.

An advantage of the seal assembly 26 is that it is slim enough to be used in conjunction with the bearings of many existing installations which were previously not provided with separate seals. The additional space required by the seal assembly is only the width of the retaining plate which in most cases need not be more than about 12 mm for a shaft of any reasonable size.

Another advantage of the seal assembly 26 is that the split ring 62 is held in place by the ring 60 and it is thus not necessary to provide other means for holding the segments together.

The hub 62 may be provided with a lug 48 which accepts a setscrew for locking the hub on the shaft.

In Figure 4 a second example of a dry seal assembly 100 is illustrated which is suitable for operation at the relatively slow rotational speeds of the shafts found in materials handling applications such as those described above. In Figure 4, the seal assembly, mounted on a shaft 101 of 64 mm diameter, is drawn substantially full size and to scale. It is not considered necessary to describe the assembly or its components in detail as these will be clear to the experienced addressee from the drawing. However, the assembly includes a stator 120 comprising a housing 102 fixed by a holding ring 104 and studs 106 to the wall 108 of a vessel. While it is commonly necessary that the housing and the rotor be of relatively expensive material such as stainless steel, the holding ring can be of less expensive material such as mild steel.

Four sealing rings 110, 112, 114, 116 are withdrawably inserted in the housing and have flanges which coact with fins 122, 124, 126 formed in the rotor 128 to define a labyrinthine passage 118 between the rotor and the stator. The rings may also be of stainless steel, or any other suitable material. The two inner rings 112, 114 are split along a diameter to enable their flanges to be inserted in the two inner annular channels defined by the fins 122, 124,

126 of the rotor 128. The respective rings 110-114 are seated in seats 130-136 formed in the housing. The rings and the respective seats in which they are located are of progressively diminishing diameter. The seats have shoulders to locate the rings in the axial direction. Through springs 137, the ring 116 is urged in the axial direction towards the fin 126. The rotor is locked on the shaft 101 by grub screws 138 mounted in a ring 140. A split locating ring 142 is fixed to the ring 140 by means of set screws 144 and, through a lip 146 received in an annular recess 148 in the stator, serves to locate the rotor 128 axially with respect to the stator 120.

An axial face lip seal 150 is mounted on the rotor 128 at its inner axial end and bears on the inner radial face of the flange of the innermost ring 110 of the stator. The lip seal 150 is of Teflon or other suitable material. The purpose of the lip seal is to provide a seal immediately after the assembly 100 is put into use. There is a large clearance between the opposing faces of the flanges and the fins as is clear from the drawing. Having regard to this large clearance, the air or other medium contained in the vessel may, in the absence of the lip seal, have a tendency to escape past the seal assembly at a velocity which prevents or slows down the build up on the faces of the flanges and fins defining the passage 118 of dust or other material which, once formed, reduces the effective width of the passage and forms the seal. The lip seal reduces the velocity of the medium through the passage. However, the solid material entrained in the medium is in most cases abrasive and soon causes the lip seal to become ineffective as a seal. However, it is intended that the life of the lip seal is sufficient to allow the build up of the material in the passage to take over the sealing action.

Run out rings 152 are inserted between the opposing faces of the flanges and the fins in the passage 118. The run out rings may be of ceramic, plastics or any other suitable material. The run out rings serve to increase the pressure drop of the gaseous medium through the passage 118 and also to locate the rings 110-116 accurately with respect to the interfacing parts of the rotor when the seal is being assembled.

Grub screws 154-160 are inserted between the housing 102 and the respective rings 110-116

to preventing the rings from rotating with the rotor. In the case of the inner rings 112, 114 the respective grub screws 156, 158 are received in axially extending slots 162, 164.

In Figure 5 a third example of a seal assembly 200 is illustrated. It is drawn in full size and to scale and is mounted on a shaft 201 of 180 mm diameter. This assembly 200 is suitable for sealing the rotating shaft of equipment which handles slurry, also at a relatively slow rotational speed. Again, in this case also it is not considered necessary to describe the assembly or its components in detail. However, it may be noted that the stator 220 comprises a split ring component 202 seated in a seat 230 machined in the wall 208 of a vessel such as a pump housing 209. The component 202 is clamped in the seat and to the wall 208 by a holding ring 204 of inexpensive material and studs (not shown) which pass through holes 206. The rotor 228 comprises three annular fins 222, 224, 226 which intermesh with three flanges 210, 212 214 machined integrally in the split ring component 202 to define a labyrinth passage 218.

A port 260 is formed in the wall 208. An oil connection or grease nipple is mounted in the outer end of the port and by this means oil or grease or water can be introduced into the passage 218.

At its outer end the component 202 is machined to receive a mechanical seal assembly 266 which will not be described in detail as it is of conventional design and construction.

In the present case the oil or grease or water in the port 260, which is at a higher pressure than the slurry in the pump housing, prevents the slurry from passing through the passage 218 and gaining access to the mechanical seal assembly 266. After the whole seal assembly has been in operation for some time, any solid material carried in the slurry which finds its way into the passage is deposited on the faces of the components defining the passage 218, substantially decreasing the effective width thereof and forming a seal of its own accord. The solid material initially entrained in the little liquid medium which penetrates to the mechanical seal is substantially removed in the labyrinth passage.

An assembly constructed substantially in the manner shown in Figure 5 may also be used to seal dry materials. In this case grease or oil may be pumped into the labyrinth passage if it is compatible with the dry material being handled. Alternatively, air at a small positive pressure may be pumped into the passage.

In both of the assemblies 100, 200 the labyrinth passages extend as much axially (i.e. parallel to the axis of rotation of the shafts) as radially.

Further examples of seal assemblies according to the invention are illustrated in Figures 6 and 7 which are also drawn to scale. Various characteristics of the seal assemblies shown in these Figures, as well as in Figures 1 to 5 and the components of which each of the seals are constructed, are given in the accompanying Table 1, which should also be referred to with reference to Figure 8.

The dimensions and other information given in Table 1 is intended to be exemplary and is not to be taken as limiting. For example, the thickness and/or the diameter of the fins in each case can be varied, usually to enable the assembly to fit into a given space. In general, the seal is likely to function more efficiently if the widths of the radial faces of the fins are kept relatively high.

Further, the diametral sizes of the components can be increased to any reasonably practical size.

The minimum particle size of the solids to be handled by the assemblies shown (and in fact seal assemblies which would be suitable for most purposes) is 3 microns.

Concerning the figures given for the rotational speed of the shaft, it should be understood that the seal assemblies would in many cases remain functional (as seals) even when the shaft is stationary. The shaft speeds are determined primarily by the apparatus (screw

conveyor etc) to which the seal assembly is fitted. Similarly, fully functional seal assemblies could be designed to fit shafts of diameter greater or less than those shown. In practice, the apparatus to which a seal assembly is fitted is in most cases unlikely to have a shaft diameter less than 25 mm.

Christopher Frederick Bayne

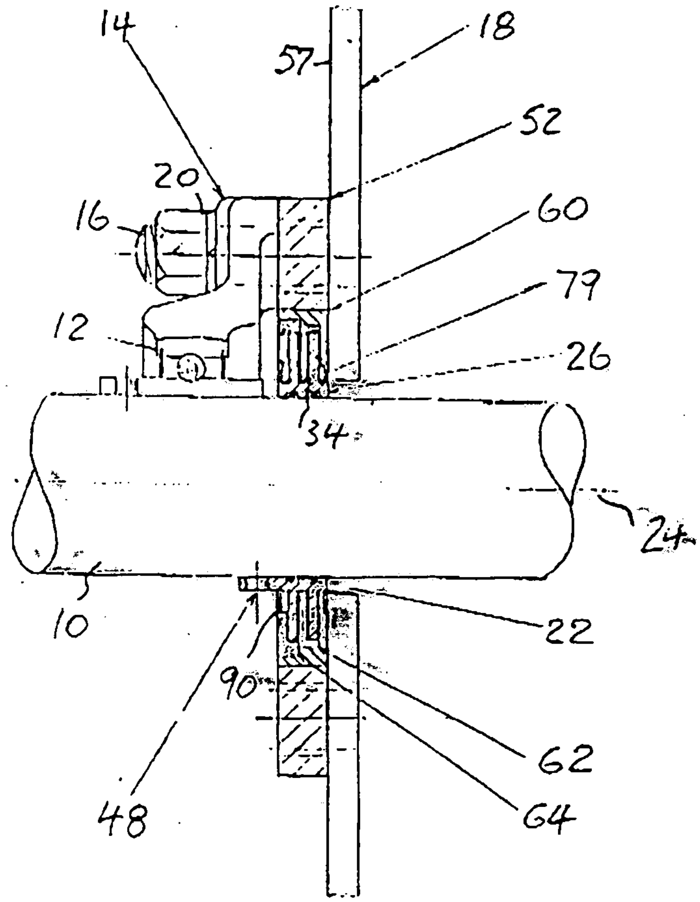
By his attorneys
PIPERS

Table 1

	Symbol in Fig 8	Ass'bly in Fig 1	Ass'bly in Fig 4	Ass'bly in Fig 5	Ass'bly in Fig 6	Ass'bly in Fig 7
Diameter of shaft	D1	50	125	180	75	50
Width of radial faces of rotor fins	W1	13.95	18.25	12	17	10.5
Width of radial faces of stator fins	W2	14	17.5	12	12.5	10.75
Thickness of rotor fins	T1	2.3	8	6.5	5	3.5
Thickness of stator fins	T2	2.3	12	5	4	2.3, 5
Outside diameter of rotor fins	D2	84.9	175	220	109	75
Inside diameter of ..stator fins	D3	57.8	147.5	200	90	54 55 58
Inside diameter of rotor fins:	D4	57	138.5	196	85	55
Outside diameter of ..stator fins	D5	85.8	183.5 193.5 201.5	224	115	75.5
Clearance between adjacent radial faces of fins in passage	C1	0.5	1	1.5	2.5	0.5
Overall length of ..passage	L	68.6	250	117	105	64.3
Rotational speed of shaft		10 - 1400	125	10-1450	5-750	10-1000
Medium to be ..handled		dry pow-der	high temp dry pow-der	slurry e.g. iron sand	dry pow-der	dry pow-der
Range of D1		25 up	25-3000	25 up	25 up	25 up

All sizes are in millimetres

Fig. 1



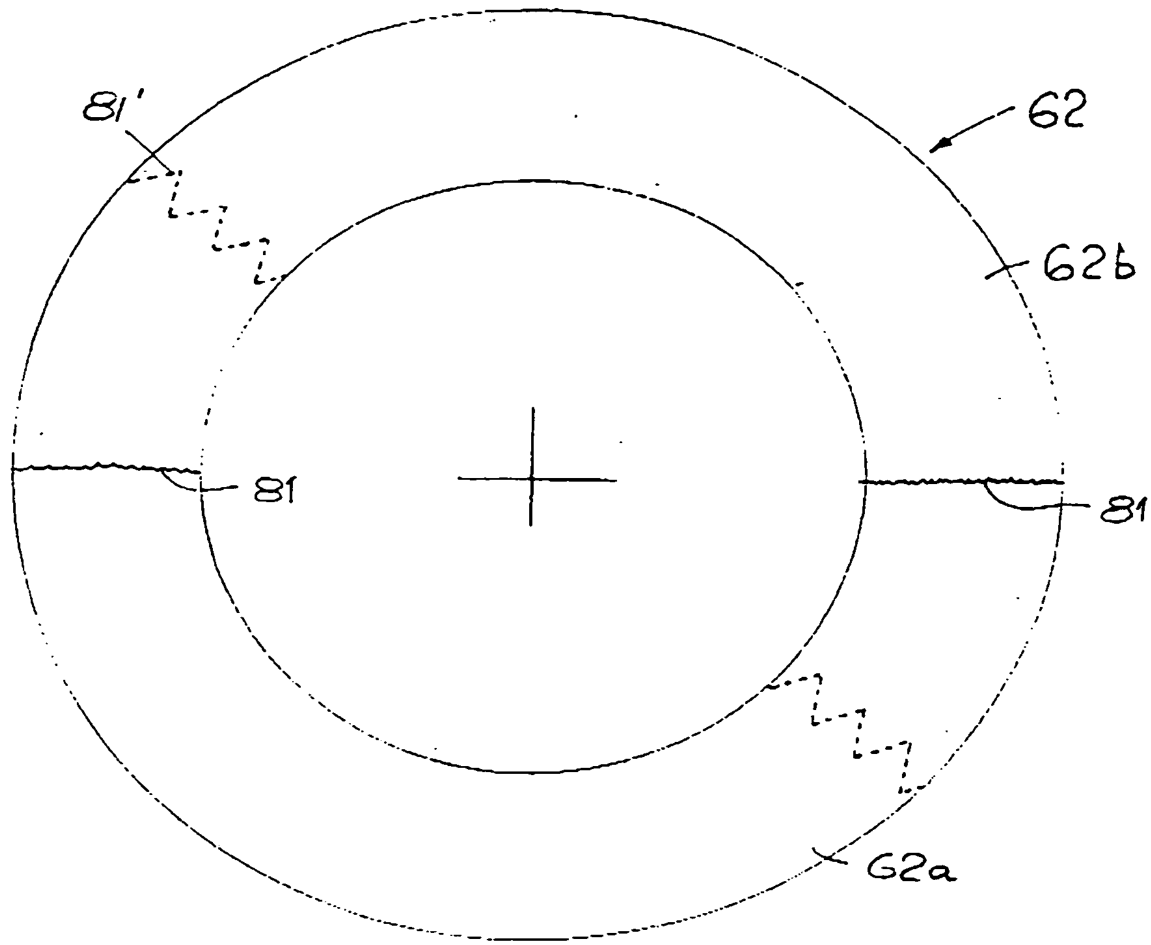
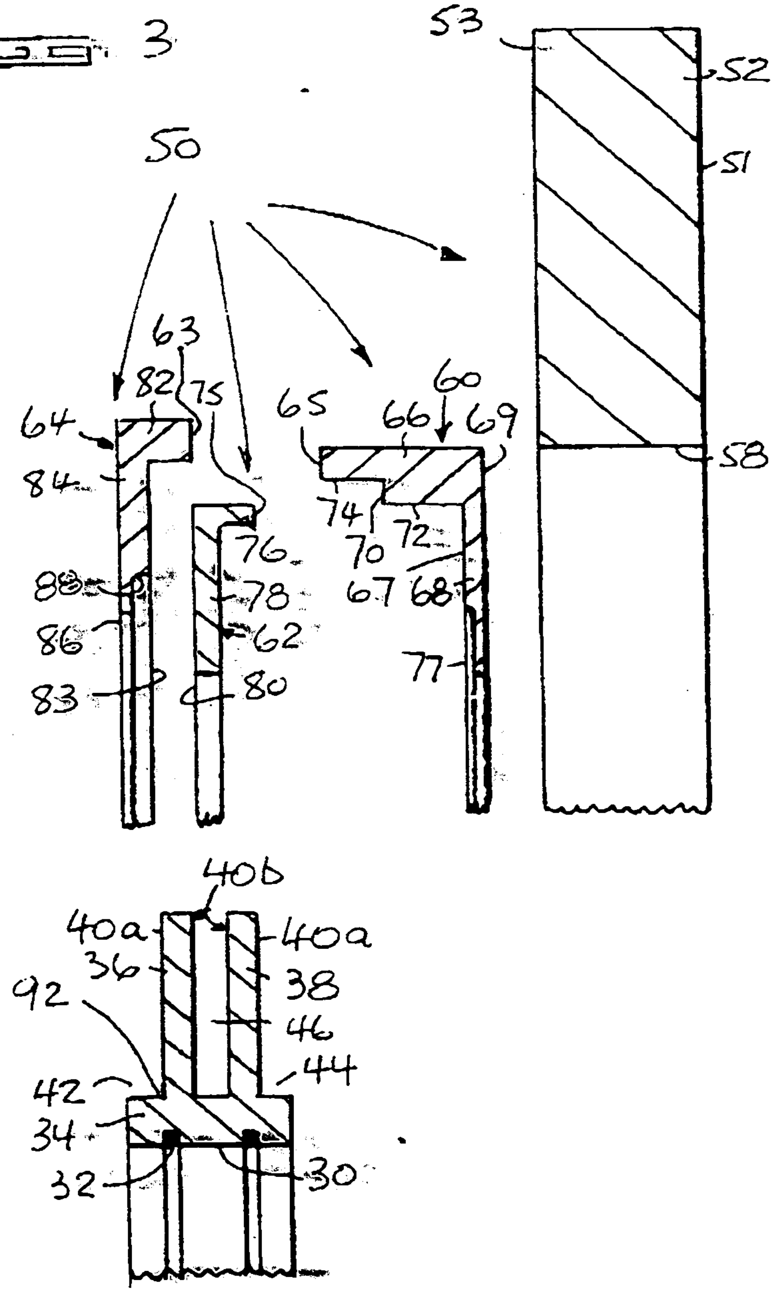


Fig. 2

FIG. 3



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83
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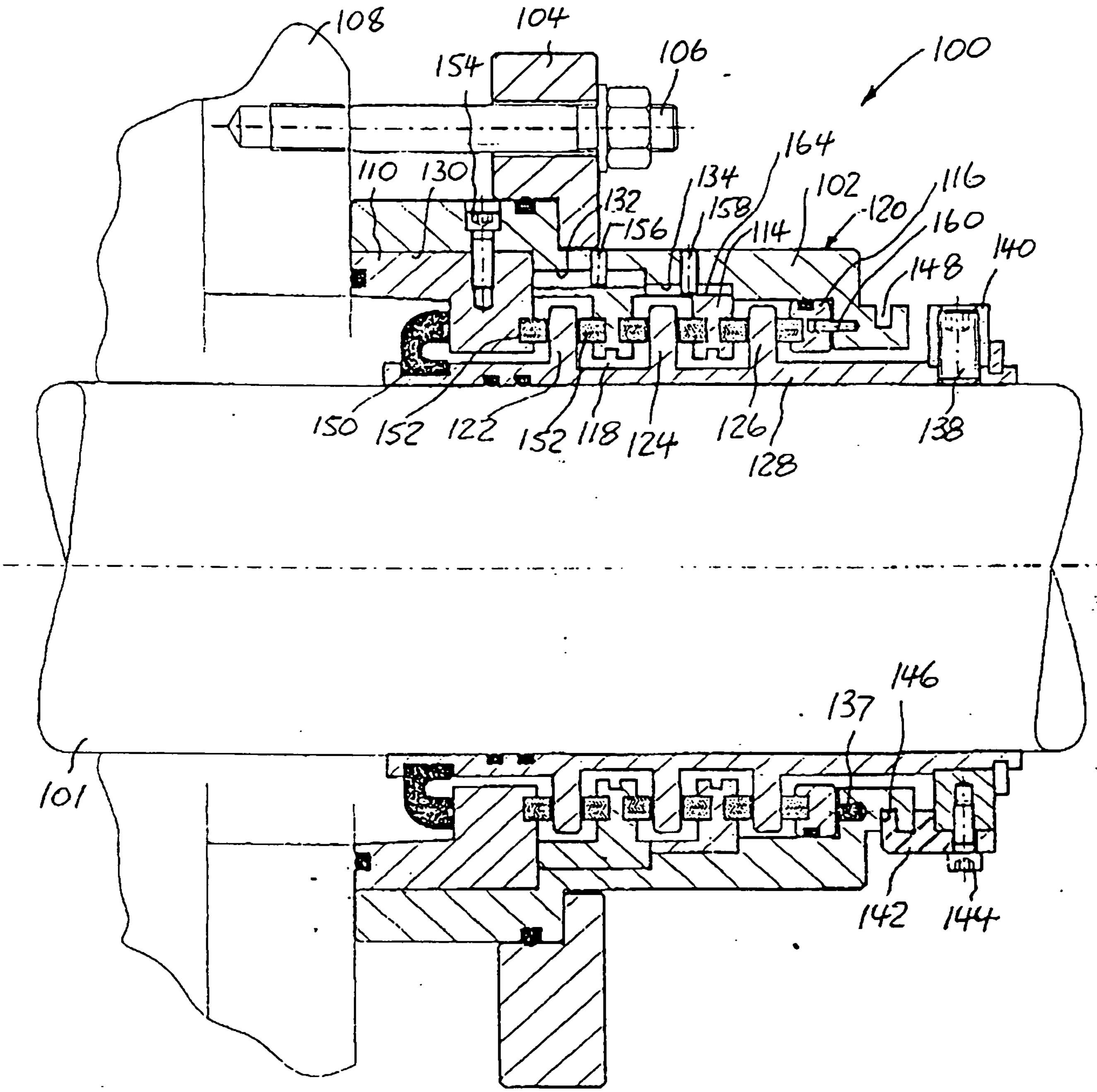
40b
40a
40a
36
38
92
46
42
44
34
32
30



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106
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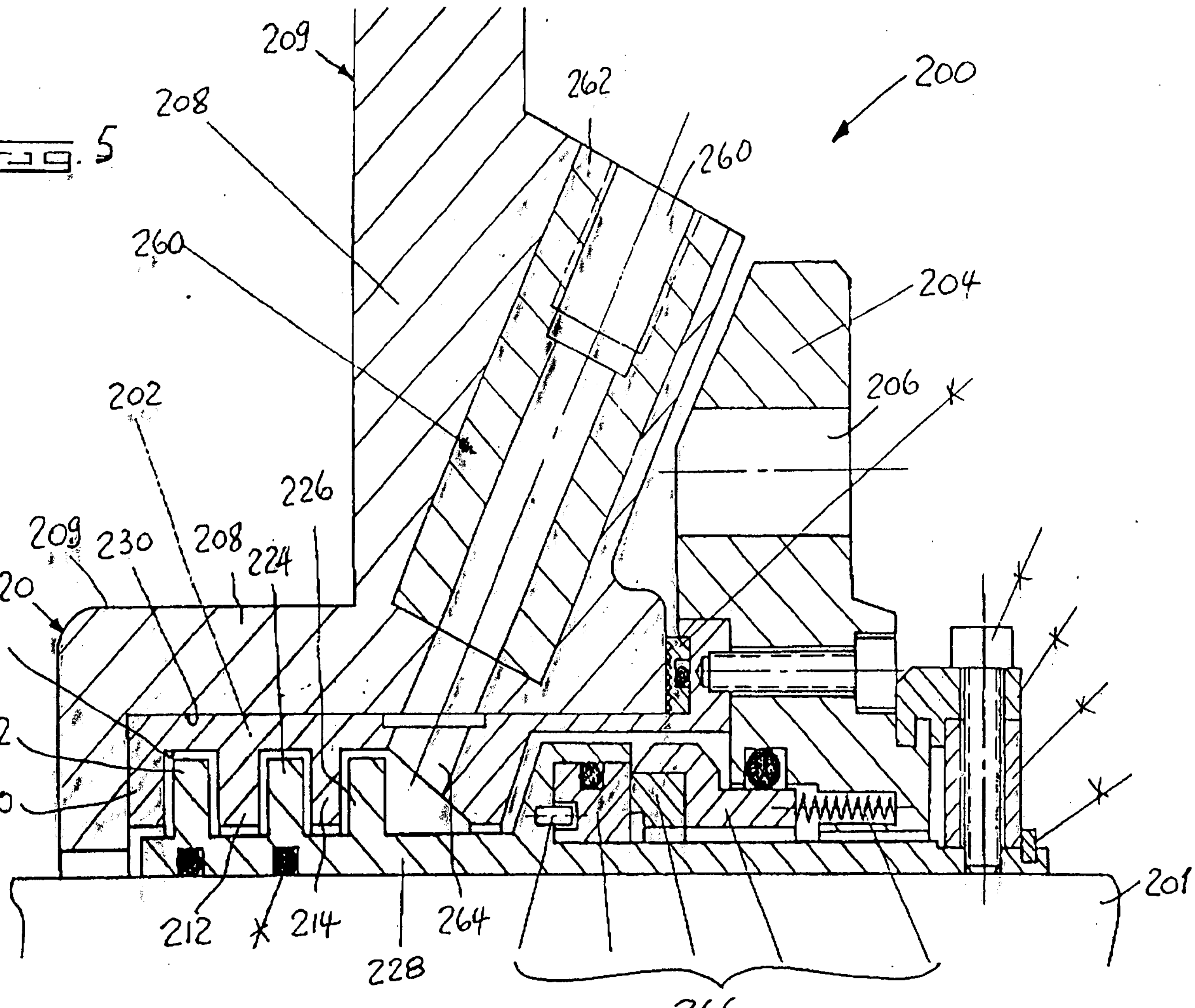
144

Fig. 4

Fig. 5



200
201
202
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222
224
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260
260
262
264
266

FIG. 6

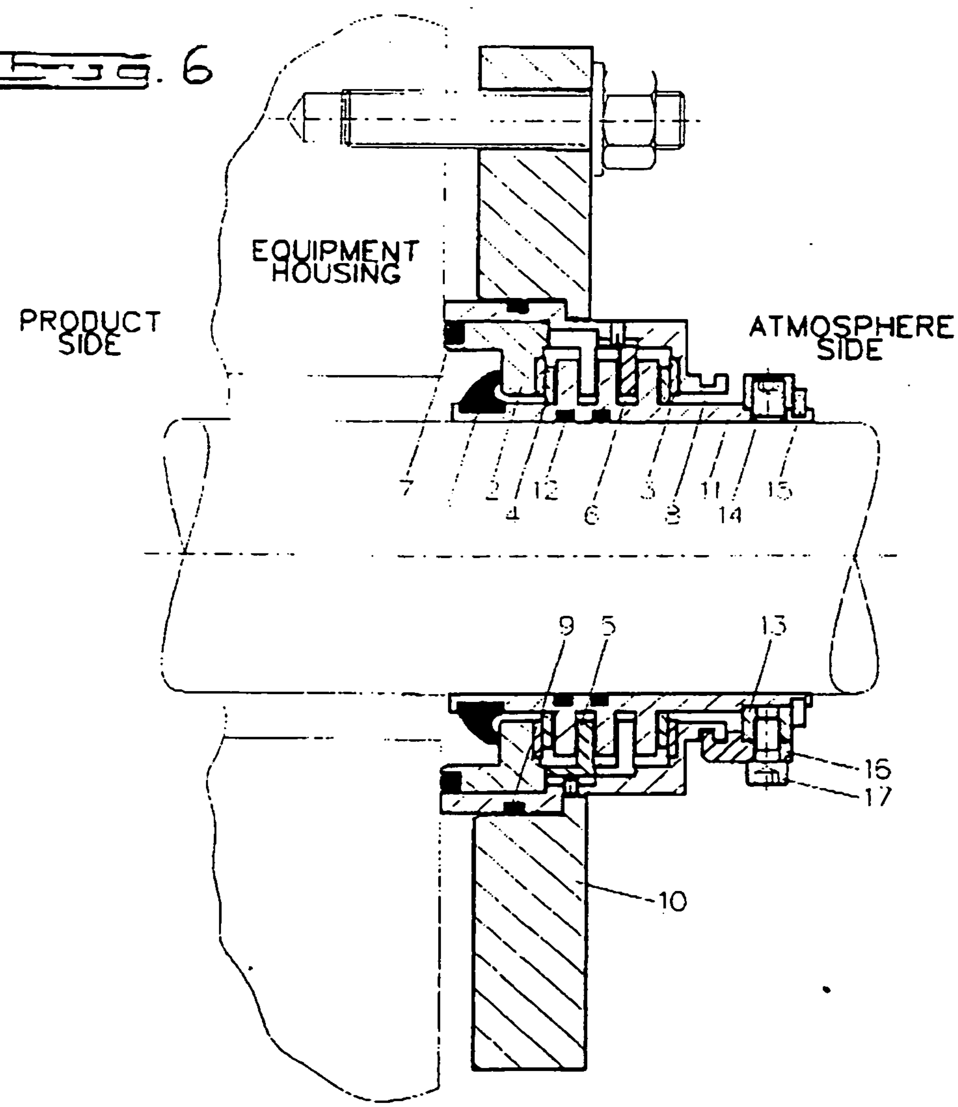
EQUIPMENT HOUSING
PRODUCT SIDE
ATMOSPHERE SIDE
7
2
12
4
6
3
8
11
14
15
9
5
13
16
17
10

Fig. 7

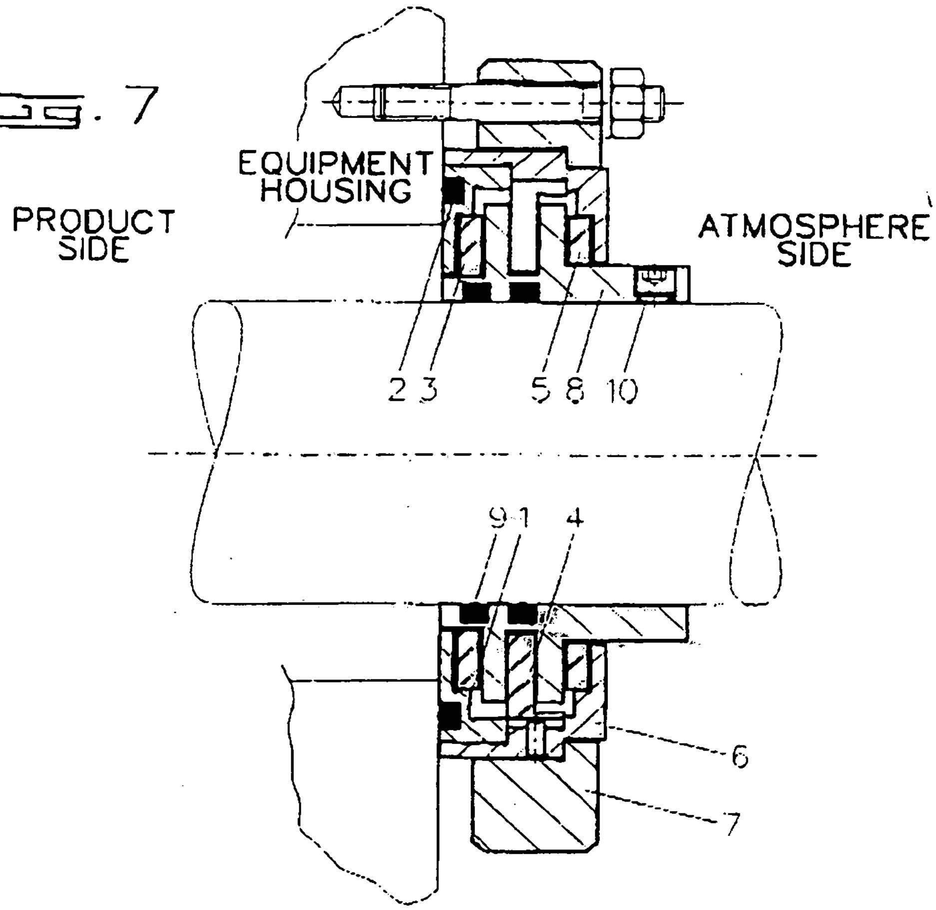
EQUIPMENT HOUSING
PRODUCT SIDE
ATMOSPHERE SIDE
2
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Fig. 8